REPORT No. 587

BLOWER COOLING OF FINNED CYLINDERS

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SUMMARY

Several electrically heated finned steel cylinders enclosed in jackets were cooled by air from a blower. The effect of the air conditions and fin dimensions on the average surface heat-transfer coefficient q and the power required to force the air around the cylinders were determined. Tests were conducted at air velocities between the fins from 10 to 130 miles per hour and at specific weights of the air varying from 0.046 to 0.074 pound per cubic foot. The fin dimensions of the cylinders covered a range of fin pitches from 0.057 to 0.25 inch, average fin thicknesses from 0.035 to 0.04 inch, and fin widths from 0.67 to 1.22 inches.

The value of q, based on the difference between the cylinder temperature and the inlet-air temperature, varied as the 0.667 power of the weight velocity of the cooling air for cylinders having spaces from 0.077 to 0.21 inch between fins. Below 0.077-inch space the exponent of the curves increased for each successive decrease in space. The value of q was independent of fin width for the range of widths tested and decreased as the space between the fins decreased.

The power required for cooling, neglecting the kinetic energy lost from the exit of the jacket, varied as the 2.69 power of the weight velocity for a given specific weight and inversely as the square of the specific weight for a given weight velocity of the cooling air. For a given weight velocity of the cooling air or a given power and for a fin width of 1.22 inches, the fin space giving the maximum heat transfer was approximately 0.045 inch.

INTRODUCTION

A general investigation is being conducted by the Committee to determine the comparative cooling of cylinders having fins of varying pitch, thickness, and width when tested in a free air stream and when tested with blower cooling. For the conditions in a free air stream the cylinders are tested with and without baffles and, for the conditions in which the blower is used, the cylinders are enclosed in a jacket.

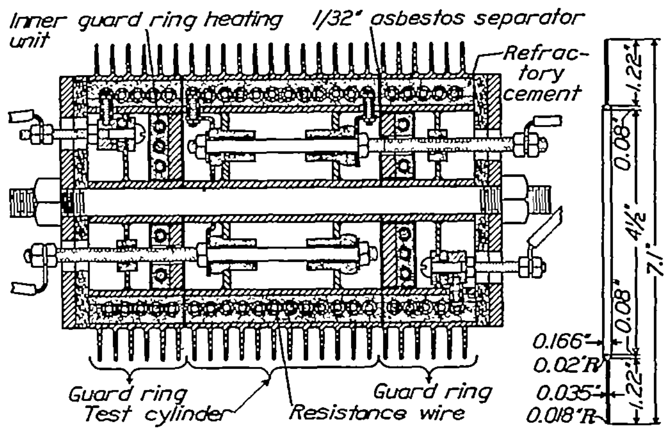
The first report published on the investigation (reference 1) presents the results of extensive tests to determine the heat-transfer coefficients of finned cylinders in a free air stream and a method for calculating the heat dissipated, utilizing these coefficients. The second report (reference 2) includes results showing how the heat-transfer coefficient may be increased by using baffles to direct the air toward the rear of the cylinder.

Tests on nine steel cylinders, herein reported, were conducted to investigate blower cooling. Tests were made on all of the cylinders to determine the effect of velocity and specific weight of the cooling air on the heat transfer and on five of the cylinders to determine the effect of the same factors on power required. The cylinders had fins of varying width and pitch; the range of fin width investigated varied from 0.67 inch to 1.22 inches, the pitch from 0.057 inch to 0.25 inch, and the thickness from 0.035 inch to 0.040 inch.

APPARATUS

TEST CYLINDER

The construction of the test unit is shown in figure 1. This unit, which has been described in detail in preceding reports (references 1 and 3), consists essentially



(a) Details of construction. (b) Individual fin section.

Figure 1.—Construction of test unit.

of three electrically heated finned cylinders, the central one forming the test section and the ones on each end serving as guard rings to prevent heat losses through the ends. The guard rings are of practically the same construction as the test section except that each ring is only one-half as long as the test section. The heat input to each guard ring and test specimen can be separately controlled by oil-cooled rheostats. A complete wiring diagram of the test set-up is shown in reference 1.

Four of the cylinders were machined from a steel billet so that the fins were integral with the cylinder wall. The other five cylinders were built up of individually constructed fins (see fig. 1 (b)) held in place by solder, a method that facilitated the making of cylinders having closely spaced wide fins. With this method of construction the same fins may be used on several cylinders of different pitch by cutting down in successive steps the thickness of the wall section and thus reducing the space between the fins. As the space is reduced, more fins are added so that the same cylinder length is maintained and the same heating unit can be used.

For convenience in referring to the finned cylinders, the designations composed of the fin pitch, width, and thickness adopted in reference 1 are also used in this report. For example, the designation 0.25–0.67–0.04 indicates a finned cylinder having a fin pitch of 0.25 inch, a fin width of 0.67 inch, and an average fin thickness of 0.04 inch. The fin proportions for each of the nine cylinders tested are shown in the following table and in figure 8.

Fin pitch (inch)	Fin width (inches)	Fin thickness (inch)	Fin space (inch)
0.25	1.22	0.04	0.21
.25	.97	.04	.21
.25	.67	.04	.21
.15	.97	.04	.11
.166	1.22	.035	.131
.137	1.22	.035	.102
.112	1.22	.035	.077
.083	1.22	.035	.048
.057	1.22	.035	.022

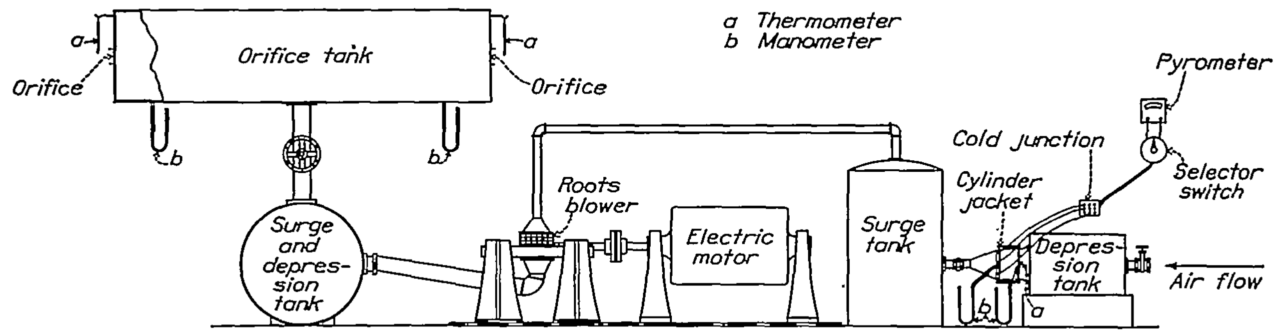
The diameter of the cylinders at the fin root was 4.66 inches, the length of the test sections 10 inches, and the length of each guard ring 5 inches.

AIR SYSTEM

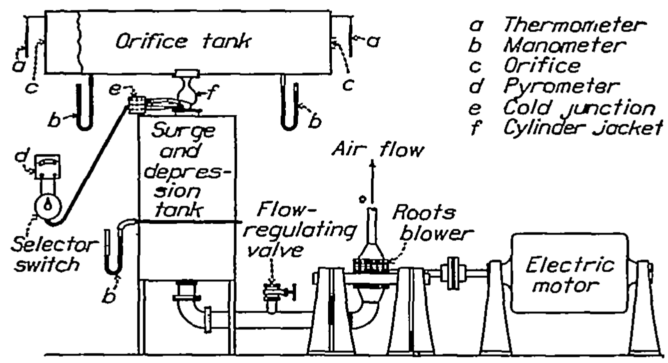
The quantity of cooling air supplied was measured by sharp-edge orifices placed at each end of a tank. The air system used in testing the 0.25–1.22–0.04, 0.25–0.97–0.04, 0.25–0.67–0.04, and 0.15–0.97–0.04 cylinders, hereinafter designated "series A" tests, is shown diagrammatically in figure 2 (a). A tank was placed in the air duct on each side of the supercharger to reduce the pressure pulsations created by the Roots blower. At the entrance of the jacket there was another tank equipped with a valve for throttling the air when the specific weight was varied.

The cooling air was directed around the cylinder by a jacket placed approximately ⅛ inch from the fin tips, as shown in figure 3 (a). Whenever the outside diameter (fin width) of the test cylinder was reduced, the ⅛-inch clearance at the tips was maintained by using sleeves inside the jacket. The inlet of the jacket was faired and proportioned in such a manner as to reduce as much as possible the breakaway of the air from the walls.

The air system used to test the 0.166–1.22–0.035, 0.137–1.22–0.035, 0.112–1.22–0.035, 0.083–1.22–0.035, and 0.057–1.22–0.035 cylinders, hereinafter designated "series B" tests, is shown diagrammatically in figure 2 (b). The jacket used on these five cylinders was in contact with the fin tips (fig. 3 (b)).



(a) Equipment used to test the 0.25–1.22–0.04, 0.25–0.97–0.04, 0.25–0.67–0.04, and 0.15–0.97–0.04 cylinders.

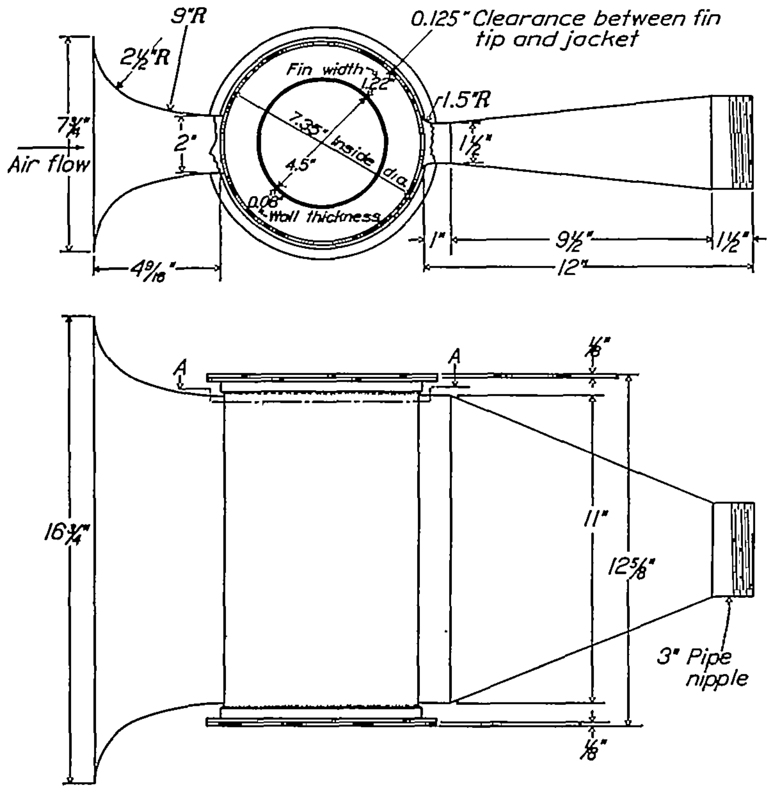


(b) Equipment used to test the 0.166–1.22–0.035, 0.137–1.22–0.035, 0.112–1.22–0.035, 0.083–1.22–0.035, and 0.057–1.22–0.035 cylinders.

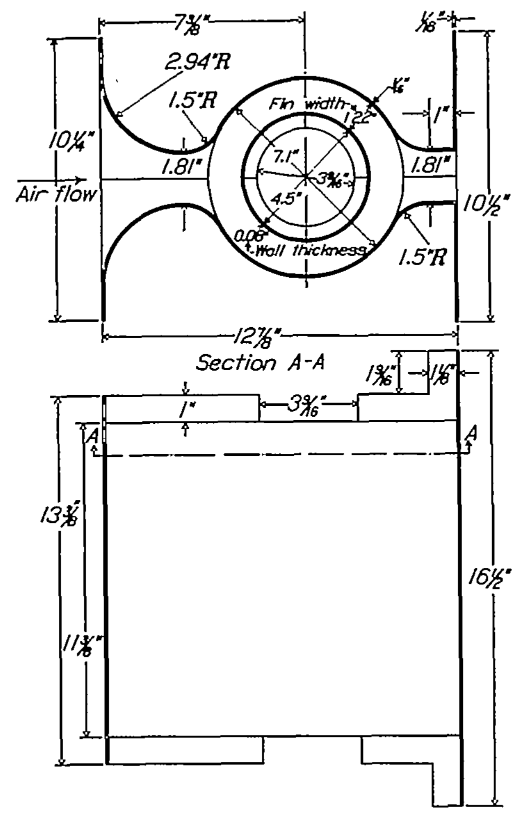
FIGURE 2.—Diagrammatic sketch of equipment.

INSTRUMENTS

The cylinder temperatures were measured with 24 iron-constantan thermocouples connected through a selector switch to a portable pyrometer. The thermo-



(a) Jacket used to test 0.25-1.22-0.04, 0.25-0.97-0.04, 0.25-0.67-0.04, and 0.16-0.97-0.04 cylinders.



(b) Jacket used to test 0.166-1.22-0.035, 0.137-1.22-0.35, 0.112-1.22-0.035, 0.083-1.22-0.035, and 0.057-1.22-0.035 cylinders.

FIGURE 3.—Sketches of jackets.

couples were made of 0.013-inch-diameter silk-covered enameled wire and were welded to the cooling surface at the points shown in figure 4. Differential thermocouples, which were connected to sensitive galvanometers, were placed on the adjacent surfaces between the guard rings and the test cylinder to facilitate adjusting the heat input to the guard rings so that there would be no heat exchange between the test section and the guard rings. Ammeters and voltmeters were used to measure the electrical power input to the test cylinder and guard rings.

The temperature of the air at the entrance of the jacket was measured with an alcohol ther-

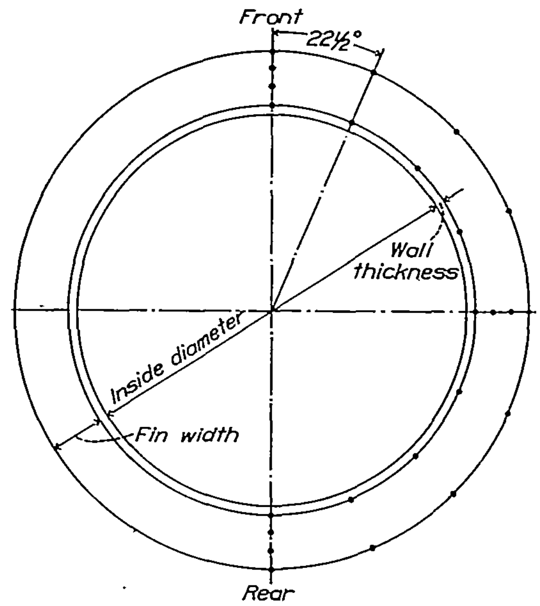


FIGURE 4.—Location of thermocouples on test cylinder.

mometer and at the exit of the jacket with three chromel-constantan thermocouples connected through a selector switch to a low-resistance portable pyrometer. In the series A tests the pressures at the entrance and the exit of the jacket and the pressure drop across the orifice tank were measured with water manometers. In the series B tests the pressure drop across the orifice tank and the pressure in the depression tank were measured with water manometers.

TESTS

Tests were conducted at air velocities from 10 to 130 miles per hour and at specific air weights from 0.046 to 0.074 pound per cubic foot. The recorded data were the electrical power input to the guard rings and test cylinder, the temperature of the air entering the orifice tank, the temperature of air entering and leaving the cooling jacket, the pressure drop across the orifice tank, the pressure at the entrance and exit of the jacket in series A tests, the pressure in the depression tank in series B tests, and the temperatures at the various points on the cooling surface.

The velocity was varied by changing the speed of the blower. The specific air weight was varied in the series A tests by throttling the air at the entrance of the depression tank. The specific weight of the air was not varied in the series B tests.

The heat inputs varied from 83 to 97 B.t.u. per square inch wall area per hour (0.0326 to 0.0381 horsepower per square inch wall area) for the various cylinders; the heat input was approximately constant, however, for any one cylinder.

The series A tests were conducted principally to determine the effect of fin width on heat transfer; those of series B were conducted to determine the effect of fin spacing on heat transfer and power required.

CALCULATIONS

The results were obtained by the following formulas:

Specific weight of the air, $\rho_1 g$:

$$\rho_1 g=\frac{1.325\times p_1}{460+T_1} \tag{1}$$

Mean velocity of the air between the fins, V_m:

$$V_m=\frac{W_t}{\rho_1 g}\times\frac{144}{A_{tr}} \tag{2}$$

(The method of calculating W_t is given in reference 4.)

Experimental and calculated heat-transfer coefficients, U_{exp} and U_{cal}:

$$U_{exp}=\frac{Q}{A_b\theta_b} \tag{3}$$

$$U_{cal}=\frac{q}{s+t}\left\{\frac{2}{a}\left(1+\frac{w}{2R_b}\right)\tanh aw'+s\right\} \tag{4}$$

where

$$a=\sqrt{\frac{2q}{kt}}$$

The value of k=2.17 for this report.

Equation (4) is derived as equation (13) in reference 1.

Average outlet cooling-air temperature T_2:

The outlet cooling-air temperature is an average of the indicated temperatures of the three thermocouples after corrections have been applied for instrument calibration and cold-junction temperature.

Power required across the test cylinder, P_t:

$$P_t=0.000893V_mA_t\{p_1-(p_2+0.00022V_m^2\rho_1 gA_t^2/A_1^2)\} \tag{5}$$

In this formula the specific weight of the air at the inlet of the jacket was used instead of the specific weights at the inlet and outlet as theoretically should be done. The error introduced by this method is small, however, and formula (5) is simpler than the rigorously correct one. It was very difficult to measure the static head at the entrance and exit of the jacket so that in formula (5) p_1 is the total head in the orifice tank (see fig. 2 (a)) and p_2 is the static head in the depression tank. The use of these heads leads to very little error unless there is a vena contracta in the entrance and exit.

Power required to generate the outlet velocity, P_a:

$$P_a=1.965\times10^{-7}A_1\rho_2 gV_2^3 \tag{6}$$

RESULTS AND DISCUSSION

The problem of blower cooling can be divided into two parts, a study of the heat transfer obtained and of the blower power required for various conditions of operation. The heat transfer for a given case can be calculated when the surface heat-transfer coefficient q of the fins is known, use being made of equation (4). A study will now be made of the dependence of q and the blower power on the fin dimensions, the physical properties of the air, and the air speed. Because a large number of variables are involved, dimensional theory is used in clarifying and simplifying the analysis.

As q depends on the specific weight, viscosity, specific heat, thermal conductivity, velocity of the air, and the various dimensions of the finned cylinder, by dimensional analysis the following expression can be set up (see equation (1), reference 1):

$$q=c_p\rho_1 gV_mf\left(\frac{\rho_1 gV_mD}{\mu},\frac{\mu c_p}{k_a},\frac{t}{D},\frac{w}{D},\frac{s}{D}\right) \tag{7}$$

With the exception of the specific heat and the conductivity of the air, the blower power depends on the same group of variables and the following relation can be obtained:

$$P_t=\rho_1 gV_m^3Df\left(\frac{\rho_1 gV_mD}{\mu},\frac{t}{D},\frac{w}{D},\frac{s}{D}\right) \tag{8}$$

where P_t is the power per unit length of cylinder. In this analysis the flow is assumed as two-dimensional, which condition the tests very closely simulated.

EFFECT OF VARIABLES ON q

Weight velocity of the air and fin dimension.—Equation (7) shows that, when all other quantities remain constant, the value of q varies as the weight velocity of the cooling air, $V_m\rho_1 g$. Tests presented herein were performed in which both the velocity and specific weight were independently varied. The values of q obtained from these tests are plotted against weight velocity on logarithmic-coordinate paper in figures 5 and 6. For any one test cylinder a straight line fitted the data fairly well.

The curves of figure 5 for the cylinders having pitches from 0.112 to 0.25 inch, inclusive, have been drawn parallel and have a slope of 0.667. For cylinders with pitches less than 0.112 inch the slope becomes increasingly greater as the pitch is decreased. From the relation between q and weight velocity shown in figure 5 for cylinders having pitches of 0.112 inch or greater, equation (7) can be modified as follows:

$$q=c_p\rho_1 gV_m\left(\frac{\rho_1 gV_mD}{\mu}\right)^{-0.333}f'\left(\frac{\mu c_p}{k_a},\frac{t}{D},\frac{w}{D},\frac{s}{D}\right) \tag{9}$$

Below 0.112-inch pitch the exponent −0.333 decreases as the pitch decreases.

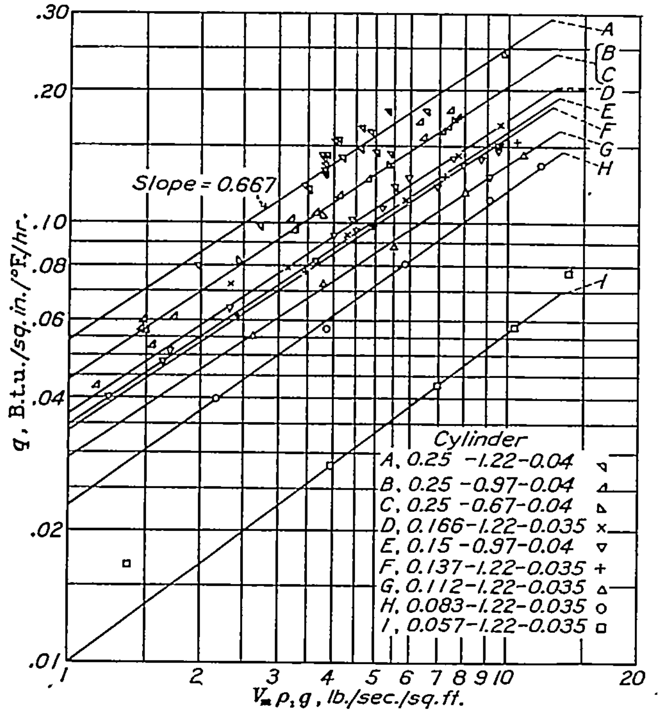


FIGURE 5.—Effect of weight velocity of the cooling air on the average surface heat-transfer coefficients, based on the difference between the cylinder temperature and the inlet-air temperature.

The curves of figure 5 in which the value of q is based on the difference between the inlet-air temperature and the average cylinder temperature show that, when the pitch is decreased, the value of q will decrease even though the weight velocity of the cooling air remains constant. If the values of q are based on the difference between the average cooling-air temperature and the average cylinder temperature, the results will be as shown in figure 6. The outlet-air temperature was calculated from the weight of air flowing over the test cylinder, the heat input to the test cylinder, the specific heat of the air, and the inlet-air temperature. It was found that more than three thermocouples in the outlet of the jacket were necessary to give a correct average temperature. Because the effect of the heating of the air on the value of q is greater at low air speeds than at high air speeds, the slope of the curves in figure 6 is much less than the slope of the curves in figure 5; all the curves in figure 6 have the same slope.

Figure 7 was obtained by cross-plotting figure 5 at a weight velocity of the air of 4 pounds per second per square foot and shows the effect of fin space on q. The surface heat-transfer coefficient varies as the 0.386 power of the fin space from 0.09- to 0.21-inch space. From 0.048- to 0.09-inch space the slope is a little greater than 0.386, and below 0.048 inch q decreases rapidly.

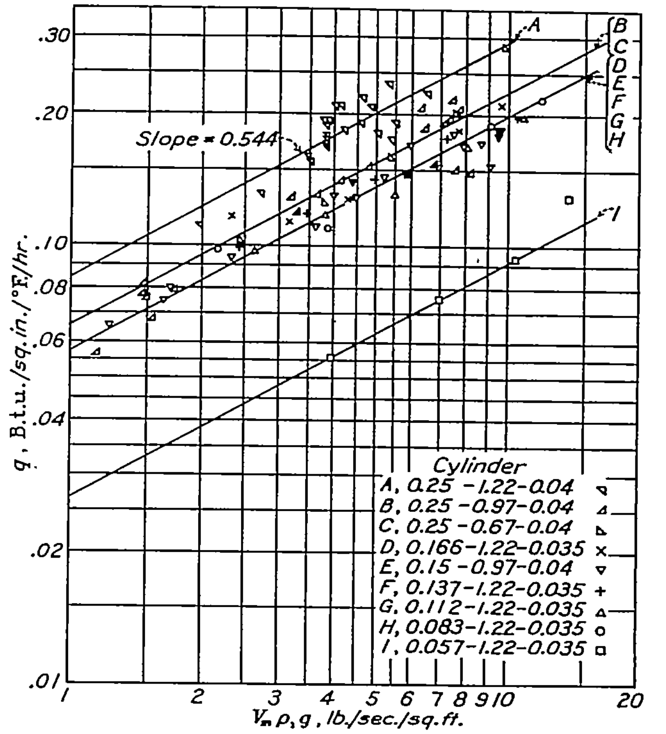


FIGURE 6.—Effect of weight velocity of the cooling air on the average surface heat-transfer coefficients, based on the difference between the cylinder temperature and the average air temperature.

It is interesting to note that the value of q, even when corrected for the heating of the air, is less for cylinders with closely spaced fins than for cylinders with widely

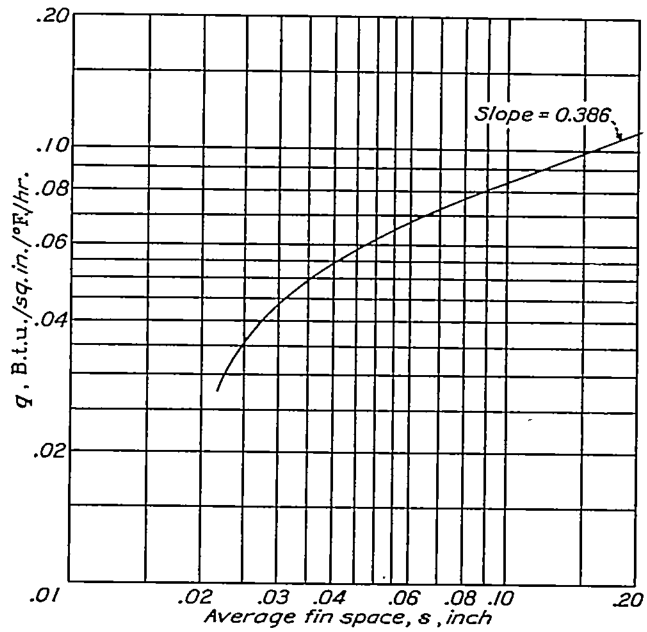


FIGURE 7.—Effect of fin space on the average surface heat-transfer coefficient. Weight velocity 4 pounds per second per square foot.

spaced fins although the average weight velocity between the fins is the same. Recent tests (reference 5) based on a study of air flow between fins indicated that the cooling was best with a 0.031-inch space between the fins, the minimum used in the air-flow tests. The cooling with closely spaced fins is greatly impaired because the flow pattern between the fins is not so conducive to a high over-all heat-transfer coefficient as the flow pattern for more widely spaced fins.

The test results indicate that fin width had little effect on q for two of the cylinders tested, the 0.25–0.97–0.04 and the 0.25–0.67–0.04. The values of q for the 0.25–1.22–0.04 cylinders are, however, greater than for the other two cylinders.

Previous tests conducted on finned cylinders having pitches of 0.15 and 0.25 inch and mounted in a free air stream indicated that, for fin widths greater than 0.4 inch, the value of q varied little with change in width (reference 1). For the tests herein reported the air was guided around the cylinder and the velocity distribution along the fin width was more uniform than for the cylinder in the free air stream (reference 5). The variation in q with fin width for two cylinders, as expected, was less than in tests on cylinders in a free air stream (reference 1). Because of the unexpected increase in q for the 0.25–1.22–0.04 cylinder, further tests are being made to determine the effect of fin width on q.

The tests on the cylinders in a free air stream also indicated that fin thickness had a minor effect on the value of q and it is reasonable to expect that the same would hold true for cylinders surrounded by a jacket. Therefore, no tests were conducted to determine the effect of fin thickness.

Air temperature.—Although no experiments were made to determine the effect of temperature of the air on q, some idea of the effect can be obtained from equation (9). The quantities u, c_p, k_a, and $\rho_1 g$ depend on the temperature of the air. The effect of $\rho_1 g$ on q has been determined. For the range of temperatures encountered in an ordinary altitude change, however, c_p, uc_p/k_a, and $u^{0.333}$ are practically constant. The heat-transfer coefficient q is therefore affected by temperature of the air only as the latter affects $\rho_1 g$.

EFFECT OF VARIABLES ON U

Weight velocity of the air and fin dimensions.—As the amount of base surface available on a cylinder for finning is limited, a fin design should be selected that gives the maximum value of U—the heat carried away per unit wall area per degree temperature difference between the cylinder wall and the cooling air per hour. Therefore, in the design of fins, the maximum cooling surface consistent with a high value of q must be used to obtain maximum cooling. The calculated values of U shown in figure 8, except for the 0.25–1.22–0.04 cylinder, were determined from equation (4) and from the values of q given in figure 5; the experimental values were computed from test results. The calculated values of U for the 0.25–1.22–0.04 cylinder shown in figure 8 were obtained from equation (4) and from the values of q shown in figure 5 for the 0.25–0.97–0.04 and 0.25–0.67–0.04 cylinders. Values of U calculated from the values of q for the 0.25–1.22–0.04 cylinder in figure 5 did not check the experimental values of U. This discrepancy is a further indication that the experimental values of q for the 0.25–1.22–0.04 cylinder are questionable and that fin width has little effect on q. These curves show that the agreement between the calculated and the experimental values is sufficiently good to justify the use of equation (4) in calculating the heat dissipated by a cylinder enclosed by a jacket.

Figure 9 is a cross plot of the experimental values of U in figure 8 and shows the effect of fin pitch on U at several constant weight velocities of the air for the cylinders with 1.22-inch fin width and 0.035-inch fin thickness. The value of U for these curves is based on the difference between the inlet-air temperature and the average cylinder-wall temperature. The values of U for the 0.112–1.22–0.035 and the 0.137–1.22–0.035 cylinders did not fall on the faired curves as well as the values of U for the other cylinders but were sufficiently close to establish this part of the curve. The calculated values of U for cylinders 0.112–1.22–0.035 and 0.137–1.22–0.035 were very close to the faired curves. The results show that for all weight velocities of the cooling air investigated the maximum heat transfer falls between cylinders of 0.057-inch and 0.083-inch pitch or 0.022-inch to 0.048-inch space. The curves have been dotted between these two values as no data were taken to establish definitely these portions of the curves. The curves show that the heat-transfer coefficient is not sensitive to the number of fins per inch for values on either side of and near the maximum. For example, with 11 or 16 fins per inch the heat-transfer coefficient U is 95 percent of the maximum value, obtained with approximately 13 fins per inch of 0.035 thickness. The fin space giving the maximum value of the heat-transfer coefficient U will vary as the fin thickness is varied and the number of fins per inch will increase as the fin thickness decreases.

The experimental values of U for the curves in figure 10 are based on the difference between the average cylinder-wall temperature and the average air temperature. The difference between the values of U in figures 9 and 10 is caused by the heating of the air. With a fin pitch of 0.05 inch and with a weight velocity of 3 pounds per square foot per second the heat-transfer coefficient is approximately 55 percent higher when based on the average cooling-air temperature; whereas, with a weight velocity of 8 pounds, the coefficient would be approximately 19 percent greater when based on the average cooling-air temperature. Likewise with a fin pitch of 0.15 inch and with a weight velocity of 3 pounds per square foot per second the heat-transfer coefficient would be approximately 29.6 percent greater when based on the average cooling-air temperature; whereas, with a weight velocity of 8, the heat-transfer

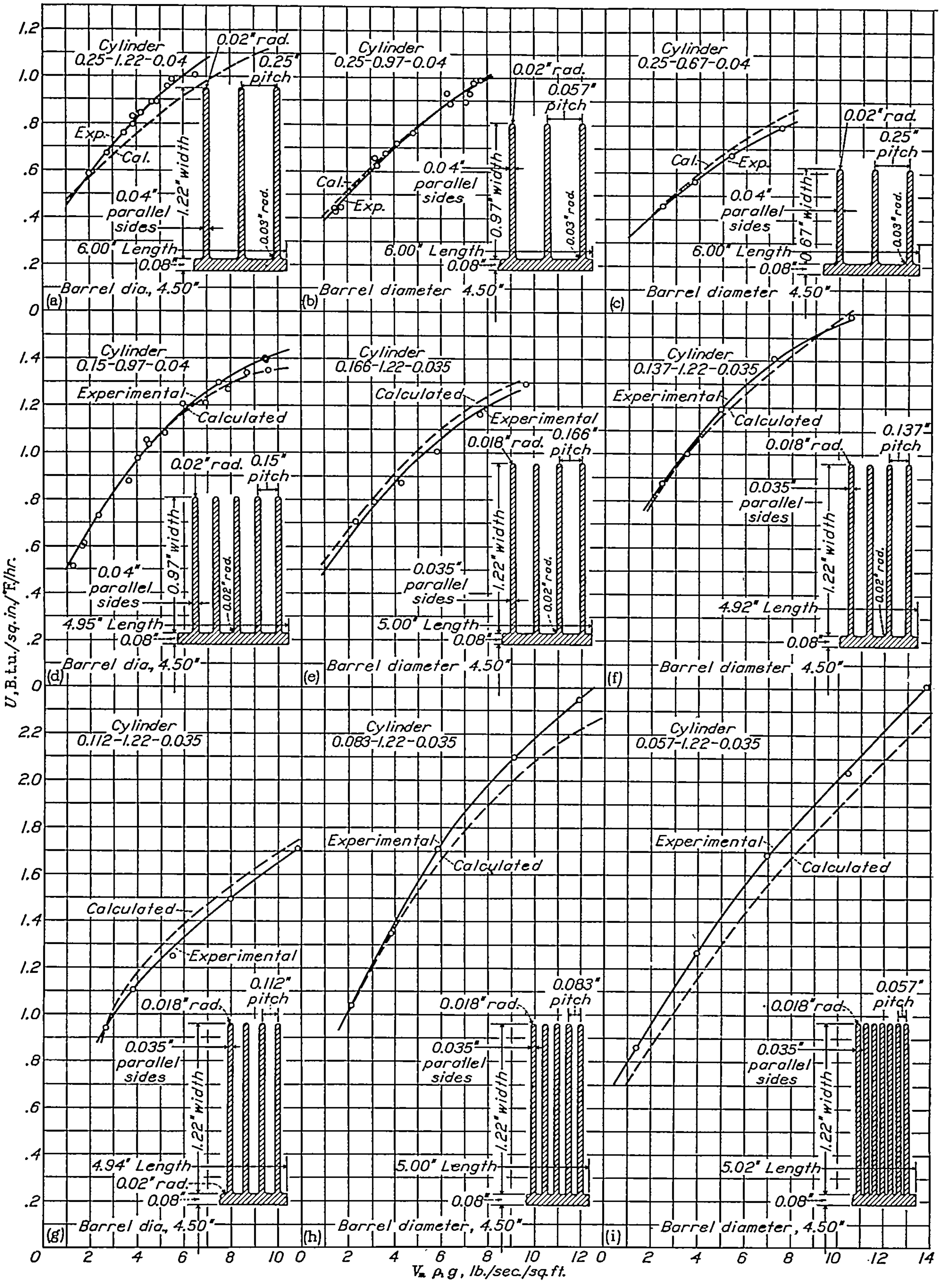


FIGURE 8.—Effect of weight velocity of the cooling air on the average experimental and calculated wall heat-transfer coefficients for the nine test cylinders.

coefficient would be approximately 17.6 percent greater. The curves in figure 10, like those in figure 9, show that the pitch for the maximum heat transfer lies between 0.057 and 0.083 inch.

EFFECT OF VARIABLES ON BLOWER POWER REQUIRED

The blower power required can be divided into two main parts: that required across the cylinder and that required to generate the outlet velocity. For a given test arrangement, the power required to generate the outlet velocity may be reduced a small amount by a properly expanding exit passage.

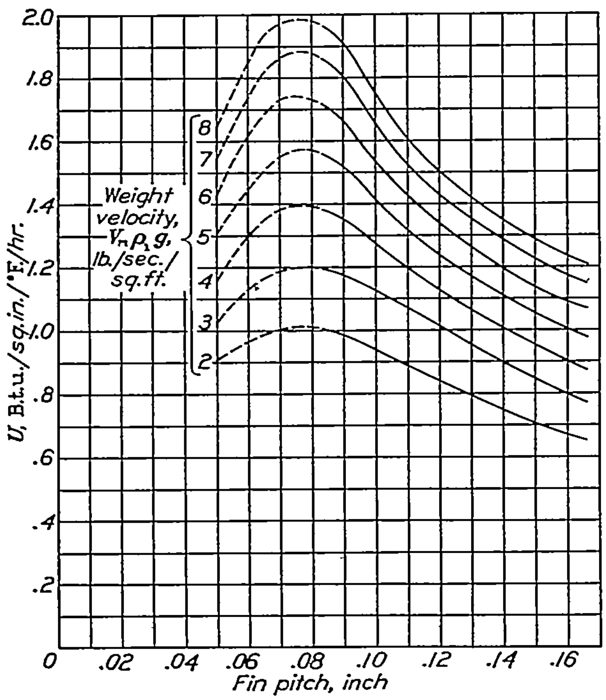


FIGURE 9.—Effect of fin pitch on the average wall heat-transfer coefficient, based on the difference between the cylinder-wall temperature and the inlet-air temperature. Fin width, 1.22 inches; fin thickness, 0.035 inch.

Weight velocity of the air and the fin dimensions.—The effect of weight velocity of the cooling air on power for five of the cylinders tested is shown in figure 11, by plotting $P_t(\rho_1 g)^2/w$ against $V_m\rho_1 g$ on logarithmic paper. The jacket around these cylinders was in contact with the fin tips, as shown in figure 3 (b). From equation (8), $P_t(\rho_1 g)^2$ varies as a function of $V_m\rho_1 g$. The effect of a small variation in the specific weight was eliminated by plotting the results in this form. Also $P_t(\rho_1 g)^2$ was divided by the fin width before plotting as it seemed reasonable to expect the pressure drop to change very little with fin width; the power would therefore vary directly as the fin width.

The slope of the curves in figure 11 shows that $P_t(\rho_1 g)^2/w$ varies as the 2.69 power of the weight velocity of the air. The data seem to show that there is a break in the curves at the lower values of weight velocity, probably caused by a change from turbulent to laminar flow but, as there are not enough points definitely to establish this break, the curves have been dotted at the lower values of weight velocity.

Dryden and Kuethe (reference 6) have shown that for flat plates the friction drag is theoretically proportional to the 1.8 power of the velocity for turbulent flow. Unpublished tests made at the Massachusetts Institute of Technology by R. H. Smith and R. T. Sauerwein show that for various finned plates the drag varied as the velocity to the 1.75 to 1.96 power, depending on the pitch and width of the fins. As the drag is directly proportional to the pressure drop in the present tests and as the power is proportional to the product of the pressure drop and the volume, the power required for friction drag should theoretically vary as the velocity to the 2.8 power, which is very close to what was obtained.

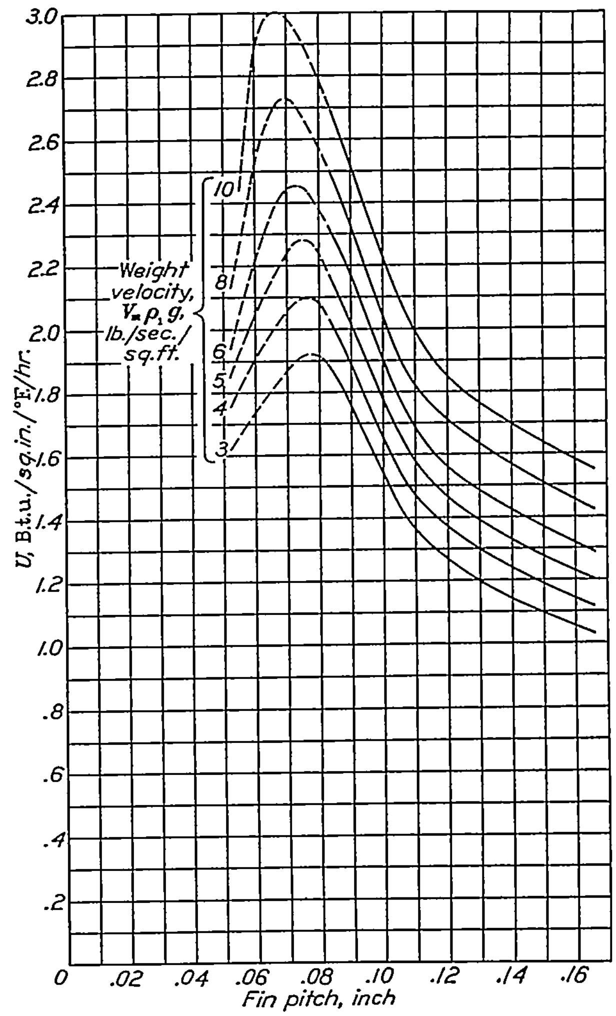


FIGURE 10.—Effect of fin pitch on the average wall heat-transfer coefficient, based on the difference between the cylinder-wall temperature and the average air temperature. Fin width, 1.22 inches; fin thickness, 0.035 inch.

From these results in order to give the observed variation of blower power with specific weight and weight velocity of the air, equation (8) must take the form

$$P_t=\frac{(\rho_1 g V_m)^3}{(\rho_1 g)^2}D\left(\frac{\rho_1 g V_m D}{\mu}\right)^{-0.31}f'\left(\frac{s}{D},\frac{t}{D},\frac{w}{D}\right) \qquad (10)$$

Figure 11 shows that the power required for cooling increases as the space between the fins decreases for the same weight velocity of the air except for the 0.166 cylinder. The data for the 0.166 cylinder fell on the same curve as the data for the 0.137-inch pitch cylinder. This result was surprising as it was expected that less

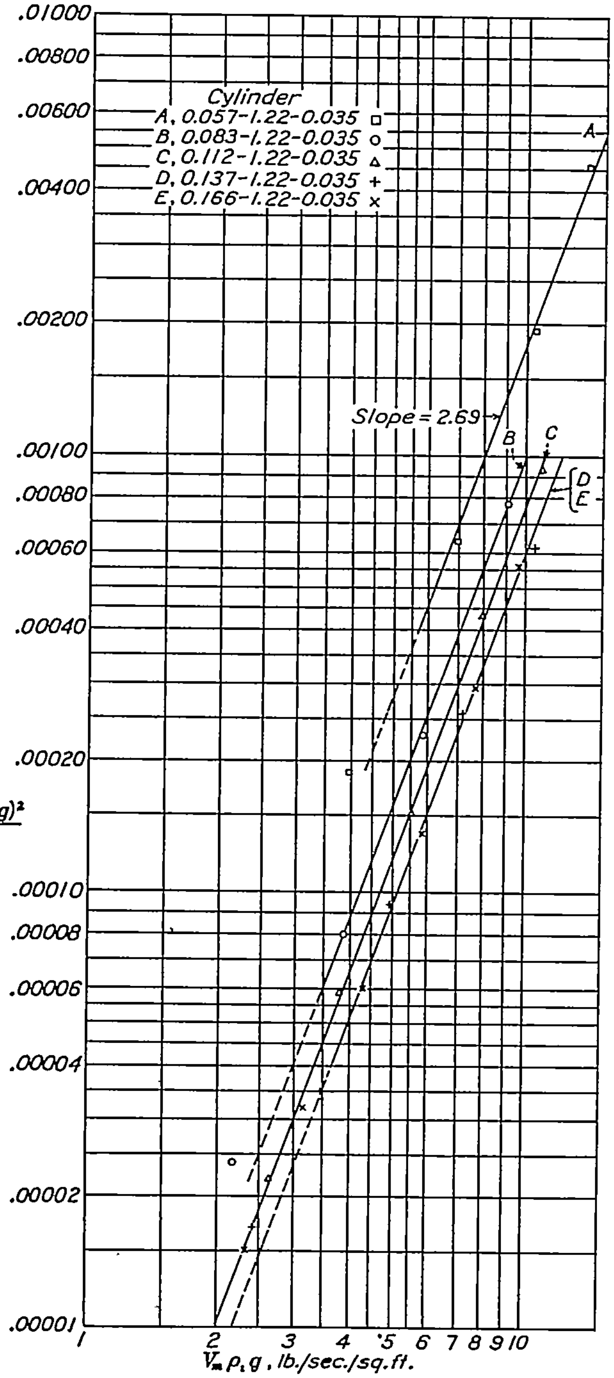


FIGURE 11.—Effect of weight velocity of the cooling air on $P_t(\rho_1 g)^2/w$.

power would be required to force air by more widely spaced fins. An analysis of the pressure drops around cylinders to be presented in a later report shows that power increases as space decreases but for the 0.166- and 0.137-inch pitch cylinders the difference is very small.

Curves of $P_b(\rho_1 g)^2/w$ plotted against weight velocity of the air are shown in figure 12 for the same cylinders as are shown in figure 11, where P_b is the total power loss across the jacket and includes both P_t and the kinetic energy lost at the exit. The total power varied as the 2.61 power of the weight velocity for all the cylinders and increased as the pitch decreased, below 0.112-inch pitch, for a constant weight velocity. The data for the 0.166, 0.137, and 0.112 cylinders are repre-

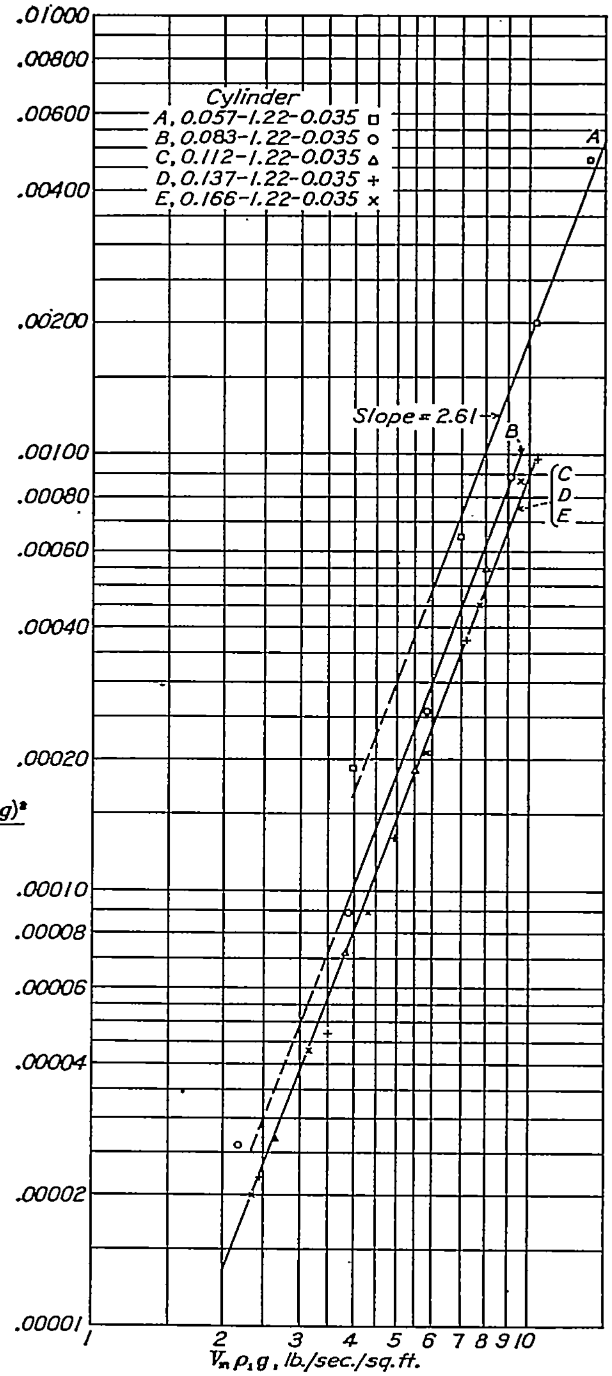


FIGURE 12.—Effect of weight velocity of the cooling air on $P_b(\rho_1 g)^2/w$.

sented by a single curve. It can be shown from figure 11 and the change in loss out the exit for the three cylinders, with a constant jacket exit area and weight velocity over the fins, that the total power required for the 0.166, the 0.137, and the 0.112 cylinders is approximately constant.

Further tests are being made to determine the effect of fin pitch, width, and Reynolds Number on the power required.

Air temperature.—The temperature of the air affects its specific weight and viscosity. The effect of variation in specific weight on power has been shown. Equation (10) shows that the power varies as the 0.31 power of the viscosity. For the range of temperatures encountered in an ordinary altitude change, the effect of change in viscosity would be small.

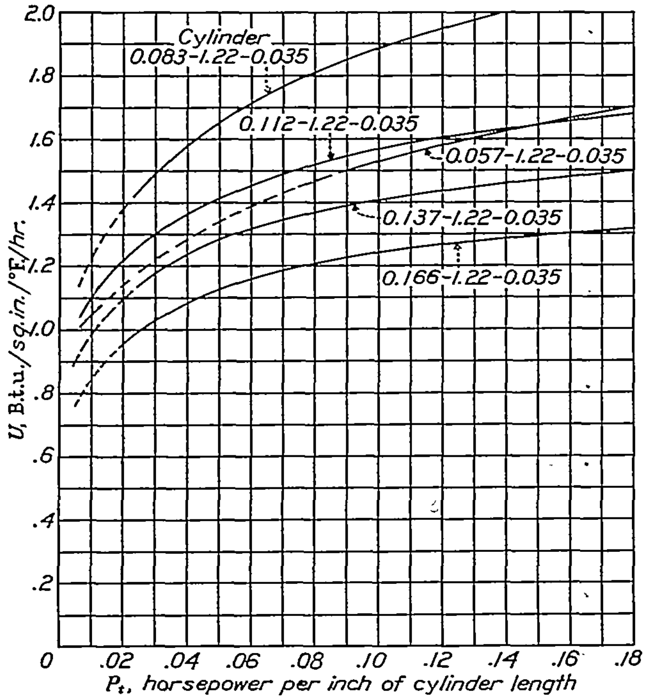


FIGURE 13.—Variation of average wall heat-transfer coefficient with power, P_t. Specific weight of the air, 0.0717 pound per cubic foot.

RELATION BETWEEN HEAT DISSIPATION AND BLOWER POWER

The wall heat-transfer coefficient U is shown plotted against the power P_t in figure 13 for a specific weight of the air of 0.0717 pound per cubic foot. These results were obtained from figures 9 and 11 and indicate that for a given power the heat transfer can be increased by decreasing the pitch up to a limiting value of approximately 0.08 inch; below this pitch the heat transfer decreases as the pitch decreases. Thus, with 0.10 horsepower, U increases from 1.24 B.t.u. per square inch per °F. per hour for the 0.166 cylinder to 1.885 for the 0.083 cylinder, an increase of approximately 52 percent, and then decreases to 1.53 B.t.u. per square inch per °F. per hour for the 0.057 cylinder. With a given horsepower, except for the 0.166-inch pitch, the weight velocity of the air decreases as the fin pitch decreases. This decrease in weight velocity tends to decrease U but decreasing the fin pitch tends to increase U until a limiting value is reached. As the effect of fin pitch predominates, the fin pitch giving maximum heat transfer for a given weight velocity will give maximum heat transfer for a given power.

Figure 14 shows curves similar to figure 13 in which the power lost as kinetic energy in the air leaving the

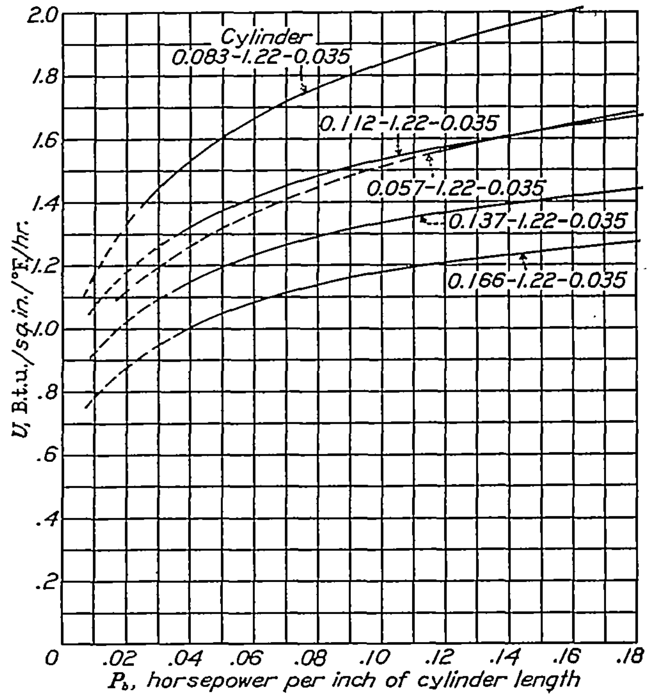


FIGURE 14.—Variation of average wall heat-transfer coefficient with total power, P_b. Specific weight of the air, 0.0717 pound per cubic foot.

exit passage of the jacket is included in calculating the required power. As previously stated, the power lost at the exit can be somewhat decreased by providing a properly expanding passage. The curves of figure 14

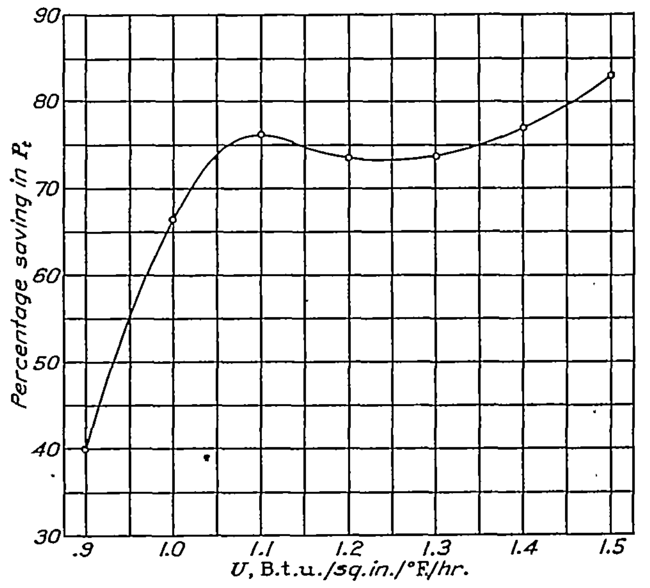


FIGURE 15.—Percentage saving in P_t by using fin pitch of 0.083 inch instead of 0.137 inch.

were obtained from figures 9 and 12 and show the same trends as do those of figure 13.

Figure 15, obtained from figure 13, is a plot of the percentage saving in P_t, the power required for cooling, through the use of a fin pitch of 0.083 inch instead of 0.137 inch at various values of U. The saving in cooling power is appreciable and, as cylinders used in conventional practice usually have fin pitches greater than 0.137 inch, it might be thought that much is to be gained from a power consideration by decreasing the pitch. The percentage of engine power required for blower cooling of conventional cylinders is, however, a small percentage of the total engine power. Löhner (reference 7) gives a value of 3.5 percent of the brake horsepower required for cooling a multicylinder engine with blowers and 8.3 percent for a single-cylinder engine. It has been found in tests of a single-cylinder engine (reference 8) that the power required for cooling varied from approximately 2.9 to 8.6 percent of the engine power, based on a blower efficiency of 70 percent and a temperature difference of 405° F. at a point between the exhaust valve and the rear spark plug, depending on cylinder and jacket design and engine-operating conditions.

CONCLUSIONS

1. The average surface heat-transfer coefficient q, based on the temperature difference between the cylinder and the inlet air, varied as the 0.667 power of the weight velocity of the cooling air for cylinders with fin spaces from 0.077 to 0.21 inch. Below 0.077 inch the exponent of the curves increased for each successive decrease in space.

2. The average surface heat-transfer coefficient q, based on the temperature difference between the cylinder and the inlet air, was independent of fin width for a range of fin widths from 0.67 inch to 1.22 inches and decreased as the space between the fins decreased. Below approximately 0.048 inch the decrease of q with fin space was very rapid.

3. The average surface heat-transfer coefficient q, based on the difference between the cylinder temperature and the average air temperature, remained constant for a given weight velocity of the air, for fin spaces from 0.048 to 0.131 inch; below approximately 0.048 inch q decreased and above 0.131 inch q increased.

4. The power required to force the air around the cylinder varied directly as the 2.69 power of the weight velocity for a constant specific weight and inversely as the square of the specific weight for a constant weight velocity of the cooling air.

5. For a given power expended in cooling, the heat dissipated from the cylinder could be increased by decreasing the space between the fins to approximately 0.045 inch for a cylinder with fins 1.22 inches wide. Below 0.045-inch space the heat dissipated decreased.

Langley Memorial Aeronautical Laboratory,
National Advisory Committee for Aeronautics,
Langley Field, Va., *November 14, 1936.*

APPENDIX

SYMBOLS

w, fin width, inches.

w', effective fin width ($w'=w+t/2$).

t, average thickness of fins, inches.

s, average space between adjacent fin surfaces, inches.

p, pitch of fins, $p=s+t$, inches.

D, cylinder diameter at fin root, inches.

R_b, radius from center of cylinder to fin root, inches ($R_b=D/2$).

R_a, average radius from center of cylinder to finned surface, inches ($R_a=R_b+w/2$).

A_b, outside base area of test cylinder, square inches (πDl).

S, total area of heated surface exposed to air stream (including fin area), square inches.

A_t, total area of spaces between fins of the test cylinder per inch of cylinder length, square inches.

A_{tr}, total area of spaces between fins of both the test cylinder and the guard rings, square inches.

A_1, area of outlet of jacket, around test cylinder per inch of cylinder length, square inches.

W_t, total weight of air flowing across test cylinder and guard rings, pounds per second.

p_1, absolute total pressure of the air in the orifice tank, inches Hg.

p_2 absolute static pressure of the air in the depression tank (fig. 2 (b)), inches Hg.

T_1, temperature of the air at the inlet of the jacket, °F.

T_2, average temperature of the air at the outlet of the jacket, °F.

V_m, average velocity of the air across the fins, feet per second.

T_b, average temperature of the root of the fin, °F.

T_m, average temperature of the root of the fin and fins of the test cylinder, °F. (These two quantities, T_b and T_m, were calculated from the test data, as explained in reference 1.)

θ_b, average temperature difference between the root of the fin and the air, °F. ($\theta_b=T_b-T_1$).

θ_m, average temperature difference between the test cylinder and the air, °F. ($\theta_m=T_m-T_1$).

Q, total heat input to test cylinder, B.t.u. per hour.

U, average over-all heat-transfer coefficient, B.t.u. per square inch base area (A_b) per hour, per °F. temperature difference between the cylinder wall and the cooling air (θ_b).

q, average surface heat-transfer coefficient, B.t.u. per square inch total surface area (S) per hour, per °F. temperature difference between the surface and the cooling air (θ_m).

c_p, specific heat of the air at constant pressure, B.t.u. per pound per °F. ($c_p=1.41\ c_v$).

μ, absolute viscosity of the air, pounds per second per foot.

k_a, thermal conductivity of the air, B.t.u. per square foot per °F. through 1 foot per second.

k, thermal conductivity of the metal, B.t.u. per square inch per °F. through 1 inch per hour.

$\rho_1 g$, specific weight of the air at the inlet of the jacket, pounds per cubic foot.

$\rho_2 g$, specific weight of the air at the outlet of the jacket, pounds per cubic foot.

V_2, velocity of the air at the outlet of the jacket, feet per second.

P_t, total horsepower per inch of cylinder length required by test cylinder to overcome losses.

P_a, horsepower required per inch of cylinder length to accelerate outlet air.

P_b, horsepower required per inch of cylinder length to accelerate outlet air and overcome all losses ($P_b=P_a+P_t$).

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